An Analysis of Solar Thermal Plant with Direct Steam Generation

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NUST Journal of Engineering Sciences, Vol.3, No.1 , Dec 2010

Abstract

Solar Thermal Power Plant Technology, also known as Concentrated Solar Power Plants (CSP), is gaining popularity due to its low cost as compared to other technologies. There are numerous technologies in CSP, however, Direct Steam Generation (DSG) is selected for fabrication cost. A study of small power generation using the said technology is studied for producing 40 W by changing various parameters. A detailed mathematical model is described for optimum selection of boiler (saturated section), super heater along with their corresponding heat losses. Mathematical calculations are further extended for bare and glasscovered tube. Analysis is performed on the basis of different Nusselt and Reynolds Number. For an open cycle, it is concluded that glasstube design is more efficient.

Introduction

Energy has become a vital part of today's technological world. Its is required in y cars, homes,industries and government offices. There has been a hike in oil prices in recent years and given the escalating demand for it, cheap energy options are becoming extinct. It has also been realized that the traditional coal / furnace oil-fired power plants have damaged the environment severely. The question is: what efforts can we make to recover or at least reduce the damage being done to our ecological system?

Therefore, there is a need to develop alternative sources of energy. Among various methods employed globally, solar electric power generation uses the sun's radiant energy to produce electricity in various ways. This can be direct as with photo-voltaic (PV), or indirect as with concentrating solar power (CSP), where the sun's energy is focused to boil pressurized water, thus replacing fossil fuels, which is then used to turn the turbine and generator to provide power.

PV is very specialized and expensive technology. Moreover, it is not environmentally friendly because it requires a very large area for its installation which damages the ecosystem. [1] Further, the PV cells manufacturing process involves the use of toxic and explosive chemicals and other health and environmental hazards which come under constant criticism from environmentalists. [2] Therefore compared to PV technology, CSP is more eco-friendly as it does not create the above issues and most importantly, *it is cheap!*

CSP offers a wide variety of methods to generate electricity,of which the Parabolic Trough method is the most effective in producing power due to its lower cost, less area requirement and design simplicity. Three options are studied in detail, namely as follows

1. Heat Transfer Fluid (HTF)
2. Direct Steam Generation (DSG)
3. Combined Power Cycle

The pros and cons of each are considered in the study and it is concluded that DSG is the most viable option for making a prototype with 40W power output. The design calculation has to incorporate all the parameters including condenser and boiler pressure, cycle selection, heat loss and area required of trough. In the following sections, this methodology is carried out for optimum area.

Mathematical Modeling of Power Plant

A. Cycle Selection

Using steam as a working fluid, its efficiency is analyzed for both open and closed cycles using ideal Rankine cycle model, with the following conditions assumed

TABLE 1
COMPARISION OF CLOSED AND OPEN CYCLE

	Closed Cycle	Open Cycle
Pressure (kPa)	101	101
Pump Inlet Quality	0.1	N/A
Pump Temperature (°C)	N/A	25
Degree of Superheat (°C)	15	15

Applying ideal conditions, we get the following graphs

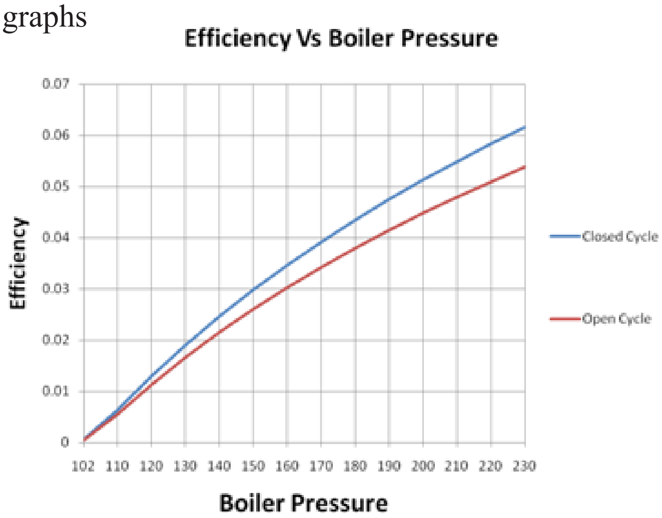

Figure 1: Ideal Rankine Efficiency comparison for open and closed cycle systems

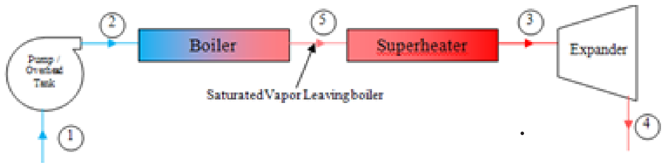

Figure 2: Schematic of Plant for Mathematical Modeling

Figure 1 shows that the closed cycle is more efficient than the open cycle and the efficiency gap widens as we move to higher turbine pressures. However, if operation is carried at low pressure, the efficiency of each cycle is very low; thus it is not practical to install a plant with low working pressure.

However, the mass flow rate required to produce 40W is quite low and it has been also observed that with increasing the pressure, the mass flow rate of steam further reduces. Therefore, for this power output, finding low flow rate pumps is difficult.

For low power output, a model steam engine, imported due to unavailability of alternatives in the local market, is available with an operating pressure of 1.4 bars. Thus, the design pressure is fixed at 140kPa. The efficiency of the plant is examined at various degrees of superheat at the same pressure, however, there is no significant difference in overall ideal efficiency of the cycle. Still, a 15 degree of superheat is fixed in order to conserve the life of the expander. The expander selected is a steam engine because turbomachines can't operate at such low flow rates. Figure 2 illustrates the schematic of model plant.

B. Coating Selection and Tube Parameters

There are two common solar selective coatings that are used for absorbing maximum possible solar radiation, which are as follows:

1. Black Chrome
2. Black Nickel

Of the two, black nickel electroplating is easily available with a maximum bath length of 1.6m. Therefore, the maximum tube length is constrained . The pipe chosen has an average diameter of 1.75in (44.45mm) with a wall thickness of 1.6mm

C. Design Calculation and Heat Distribution

The power output is fixed to 40W and the expander pressure is restricted to 140kPa. It is assumed that the engine isentropic efficiency is 70% where as the pumping efficiency is 80%. The mass flow rate for this cycle under the specified conditions is calculated to be 1 g/s and the total input energy required by the fluid is 2.655kW.

$$\dot{Q}_{superheater} = \dot{m} \cdot (h_3 - h_5) \tag{1}$$

From this, 31.3W is required to superheat by 15 degrees, while the rest of it is required to produce saturated steam at the flow rate mentioned above.

D. Determination of Surface Temperatures

It is also calculated that the entry length of the flow in the super-heater is greater than the total length of the tube itself. Therefore, the heat transfer co-efficient in it is calculated using the relation of the Nusselt Number [3] for developing flow against increasing lengths

$$Nu = 3.66 + \frac{0.065 \cdot \left(\frac{d_i}{L_{superheater}}\right) \cdot Re \cdot Pr}{1 + 0.04 \cdot \left(\left(\frac{d_i}{L_{superheater}}\right) \cdot Re \cdot Pr\right)^{\frac{2}{3}}} \tag{2}$$

It is calculated that the heat transfer coefficient decreases marginally after increasing length of the superheater. However, there is a drop in surface temperature since the inside surface area of the superheater is increased while keeping the input energy fixed at 31W. The relation used for the above purpose is

$$T_{surface} = \frac{Q_{superheter}}{h \cdot A} + T_{steam} \qquad (3)$$

Figure 3 shows the variation of surface temperature of the superheater against increasing lengths.

In the boiler section, the effective heat flux incident on pipe surface being transferred to the fluid is calculated using the following relation, assuming fluid in pipe receives heat uniformly from all directions

$$\dot{Q}_{boiler} = \frac{\dot{q}_{boiler,flux} \cdot \pi \cdot d_o \cdot L_{boiler}}{2}$$

which is calculated for 0.10m super-heater length to be 24.18 kW/m2, and the surface temperature is calculated using the relation [4]:

$$\dot{q}_{flux} = \mu_l \cdot h_{fg} \cdot \left[\frac{g \cdot (\rho_l - \rho_v)}{st}\right]^{\frac{1}{2}} \cdot \left[\frac{C_{pl} \cdot (T_{s,boiler\ inside} - T_{sat})}{C_{sf} \cdot h_{fg} \cdot Pr_L^n}\right]$$

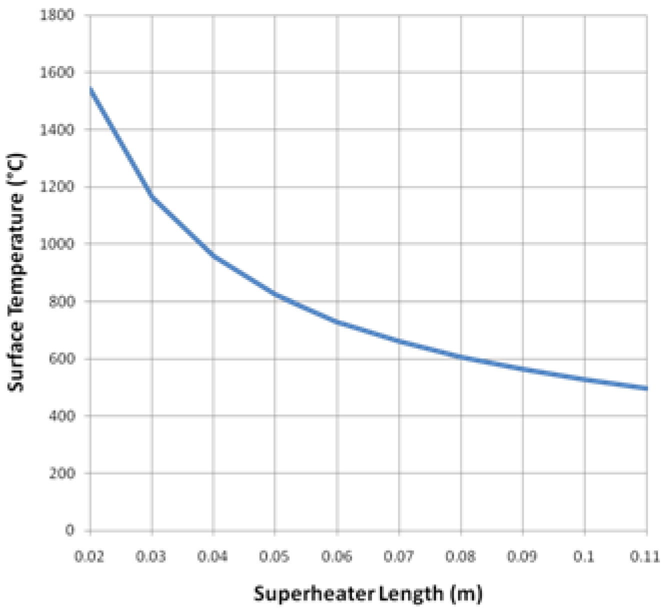

Figure 3: Variation of Surface Temperature against Superheater Length

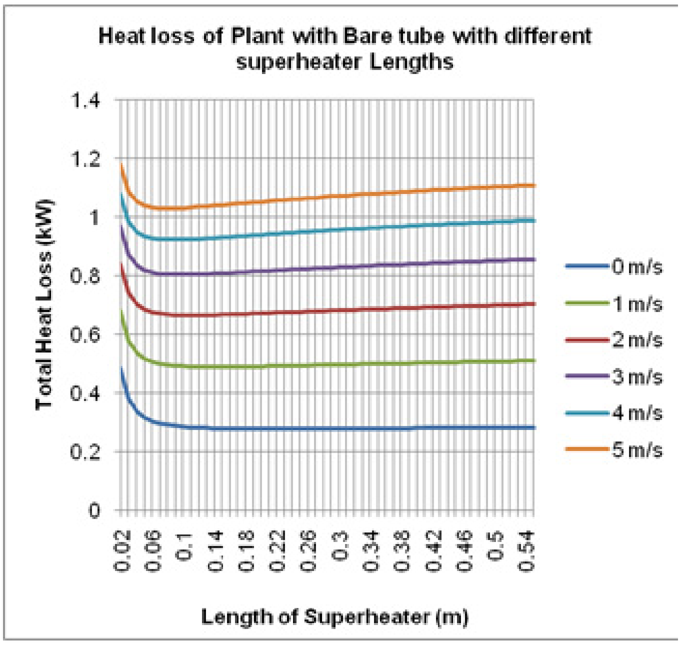

Figure 4: Total Plant Heat Loss at Different Wind speed with bare tube

The surface temperature is found to be 3 degrees higher than the saturation temperature for scored surface and 5.5 degrees higher for extremely polished surfaces. However, the pipe given is a scored one, therefore, the corresponding surface temperature is selected.

Heat Loss Analysis

A. Bare Tube Analysis

Previously, it was calculated that increasing the length of the super-heater, decreased its surface temperature, and thus together, the heat losses are decreased. However, on further increasing the length, the surface area factor dominates and the heat losses increase. Also, as the wind velocity increases, so does the heat loss. Figure 4 demonstrate the trend explained above.

One thing to note is that the minimum heat loss for bare tube also varies with the wind speed as shown in the following table:

TABLE 2
MINIMUM HEAT LOSS AT DIFFERENT WIND SPEEDS AND CORRESPONDING LENGTH OF SUPERHEATER

Wind Speed (m/s)	Minimized Heat Loss (kW)	Length of Super-heater (m)
0	0.2788	0.22
1	0.4903	0.14
2	0.6665	0.11
3	0.8051	0.0905
4	0.9244	0.0805
5	1.032	0.08

B. Glass Analysis

Following assumptions are made in the glass tube analysis:

1. The glass tube has a diameter of 2" (50.8 mm)
2. There is no vacuum inside the glass tube, air inside is at 1 atm.
3. The surface temperature of the glass is equal from inside and outside due to low thickness.
4. The heat lost by the absorber tube is in the form of convection and radiation and the radiated heat is completely transmitted through the glass.
5. The convective loss from the copper pipe is then transferred to the glass where it is conducted followed by heat loss to the surrounding atmosphere via convection and radiation.
6. The emissivity of glass is 90%.
7. The glass tube and the absorber tube have the same length.

The natural convection loss between glass and the tube is calculated using the following relations [5]

$$F_{cyl} = \frac{\left[ln\left(\frac{D_g}{D_o}\right)\right]^4}{L_c{}^3 \cdot \left(D_o{}^{-\frac{3}{5}} + D_g{}^{-\frac{3}{5}}\right)^5} \qquad (6)$$

$$k_{eff} = 0.386 \cdot k_{gb} \cdot \left[\frac{Pr_{gb}}{Pr_{gb} + 0.861}\right]^{0.25} \cdot \left(F_{cyl} \cdot Ra_{gb}\right)^{0.25} \qquad (7)$$

Where L_c stands for characteristic length, D_o represents the outside diameter, the subscript 'gb' stands for air between glass and boiler, Ra donates Rayleigh Number where as Pr donates Prandtl number [5].

$$\dot{Q}_{lost,conv,gb} = \frac{2 \cdot \pi \cdot k_{eff}}{\ln\left[\frac{D_g}{D_o}\right]} \cdot \left(T_{sb} - T_g\right) \qquad (8)$$

The heat loss via radiation between glass and super-heater is calculated by [6]:

$$\dot{Q}_{lost,rad,gb} = \frac{\sigma \,.\, A_{s,gb} \cdot \left(T_{sb}{}^4 - T_g{}^4\right)}{\frac{1}{\varepsilon} + \left[\frac{1-\varepsilon_g}{\varepsilon_g}\right] \cdot \left[\frac{D_o}{D_g}\right]} \qquad (9)$$

Finding the actual heat loss is an iterative procedure because value of glass surface temperature is not known. The iterations are stopped when the heat loss by convection between boiler surface and glass equals the total heat loss by glass tube to the atmosphere.

Figure 5 shows that it is calculated that for a fixed length of super-heater, by using glass tube the heat loss is reduced by a considerable level and becomes almost independent of the wind speed whereas for the bare tube it increases almost linearly with the wind velocity

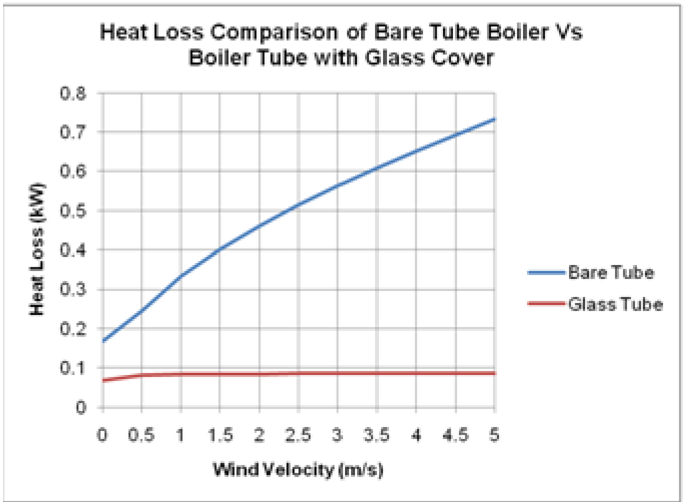

Figure 5: Heat Loss Comparison of Bare Tube Boiler and Boiler Tube with Glass Cover

Similar mathematical model is applied to super-heater section against its various lengths. It is calculated that increasing the length of the super-heater decreases the super-heater heat loss for glass whereas it is opposite for bare tube after some particular length (Figure 6)

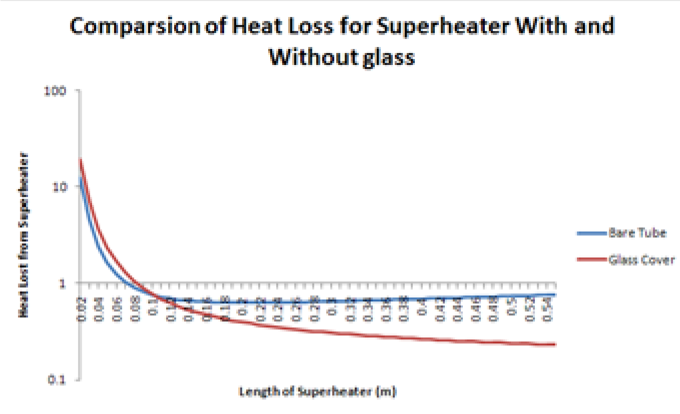

Figure 6: Heat Loss of super-heater with and without glass at wind speed of 5 m/s

Then the total heat loss at constant boiler pressure (140 kPa) is determined by summing the super-heater and boiler heat loss. As expected, the use of glass tube significantly reduced the total plant heat loss and more importantly, it is almost independent of wind speed. (Figure 7)

Area Required

It is calculated that during the average solar heat flux

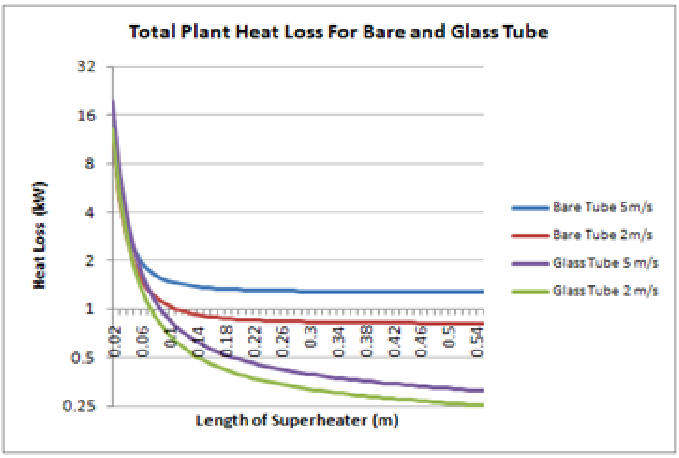

Figure 7: Total Plant Loss with and without glass

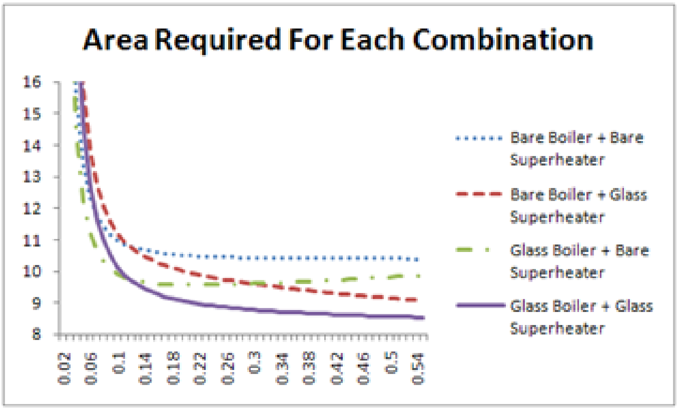

Figure 8: Area required for different combinations of Tube Selections

incident at Karachi is 0.456 kW/m2 during the 12 hours of the day, and the results are approximately close to the world solar energy map.

The required area for the parabolic collector is calculated by adding the total heat loss and the heat absorbed by the fluid and dividing it by the absorber efficiency, mirror reflectance efficiency and glass transmittance (if used) which are assumed to be 90%, 90% and 97% respectively. The results for each combination are compared and it is observed that the least area is required when both the super-heater and boiler are covered with glass tube. (Figure 8)

Although, increasing the length of the super-heater decreases the total area, however, this overshadows the fact that the width of the parabola for the super-heater and boiler would be different for each region, which will be difficult to control in construction and movement.

In order to calculate the width of parabola, the area of each section, boiler and superheater, is divided by the corresponding length and the results are shown in Figure 9.

As evident from the graph that in order to have a uniform parabola, least width is required with combination of glass super-heater and glass boiler at the length of 0.12m super-heater.

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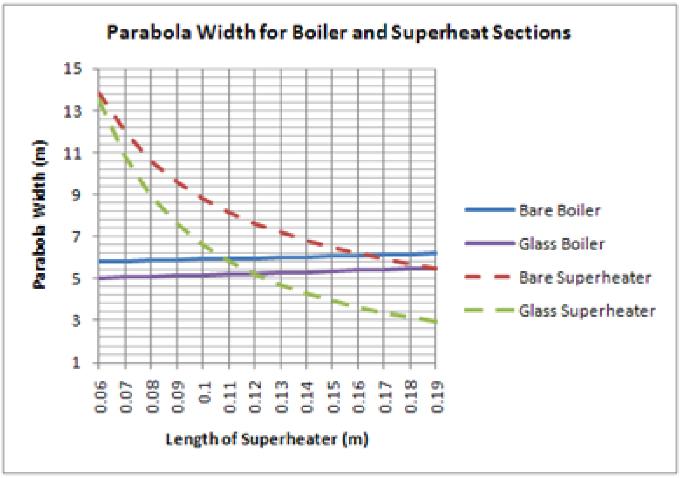

Figure 9: Variation of Parabola Width with different combinations against Super heater Length

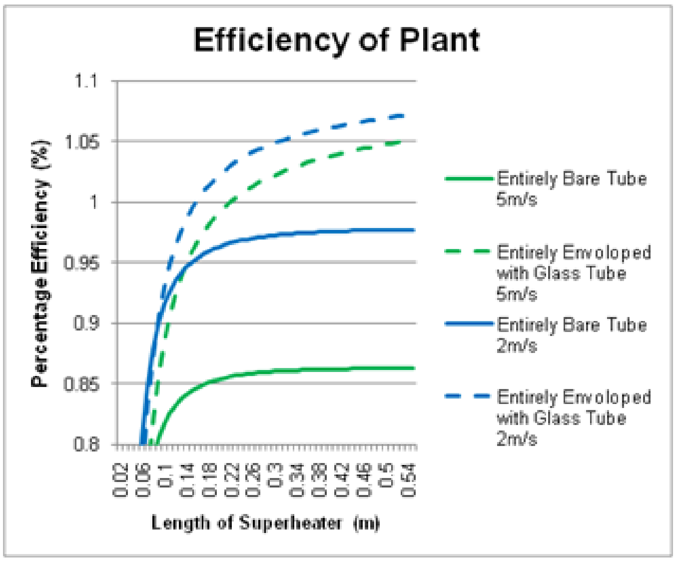

Figure 10: Total Plant Efficiency

Efficency Of Plant

The Carnot efficiency of the plant is evaluated using the formula:

$$\eta_{carnot} = 1 - \frac{T_{sat@140kPa}}{T_{sat@100kPa}} \qquad (9)$$

It is calculated to be 2.36%.

The actual thermal efficiency of the plant is evaluated using general formula of

$$\eta_{th} = \frac{\dot{W}_{net,out}}{\dot{Q}_{solar}} \qquad (10)$$

The results are calculated, as usual, for different lengths of the super-heater and the results are shown in Figure 10.

At our optimized super-heater length, the total plant efficiency is evaluated to be 1.07% with glass and 0.918% without it.

Analysis at different pressures

The behavior of the optimized design is then analyzed at different pressures and the results show a significant improvement.

A. Super-heater Surface Temperature

As the pressure increases, the surface temperature of super-heater decreases. This is because of the increase in density and the thermal conductivity of steam at higher pressures which increase the heat transfer coefficient and thus decrease the surface temperature. Figure 12 explains this behviour

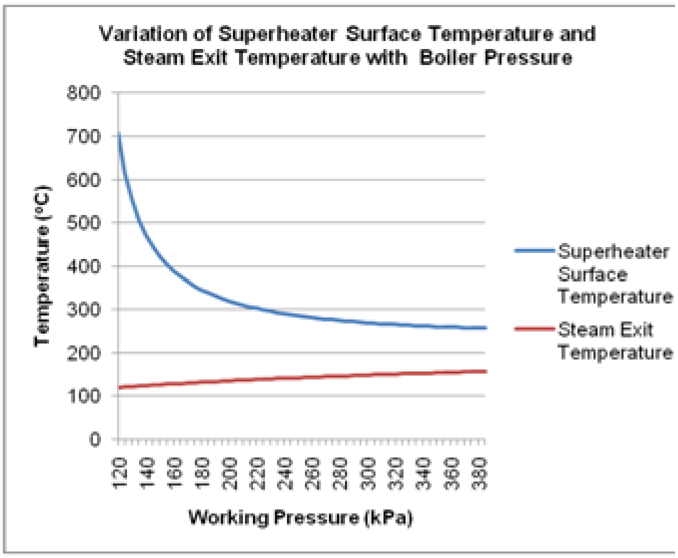

Figure 12: Super heater Pipe Surface Temperature against Increasing Pressure

B. Total Heat Loss of the Plant

As the super-heater surface temperature decreases on increasing the pressure, the heat loss decreases initially. On increasing the pressure, the Boiler surface temperature increases due to the rise in saturation temperature. This further increases the saturation temperature of the boiler section and in turn increases the heat loss. Please refer to Figure 13 for details.

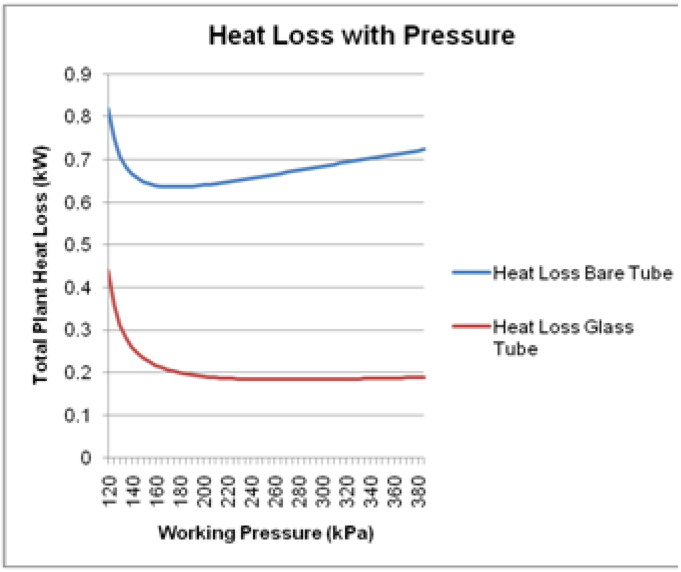

Figure 13: Total Plant Heat Loss with Increasing Pressure

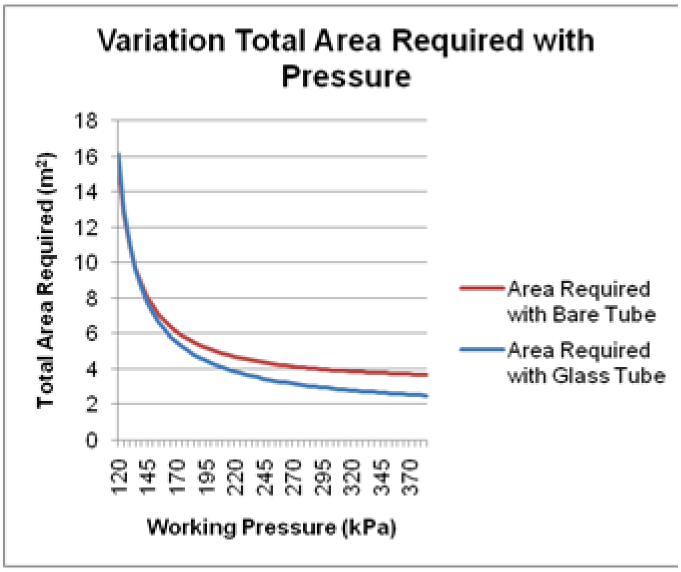

Figure 15: Area Required against Increasing Pressure

C. Thermal Efficiency of the Plant and the Area required

The thermal efficiency of the plant increases with increasing pressure due the rise in saturation temperature of the fluid and thus the Carnot efficiency and thermal efficiency which is less than the thermal efficiency (Figure 14)

The increase in thermal efficiency leads to reduction in the total area required to produce the same amount of power. (Figure 15)

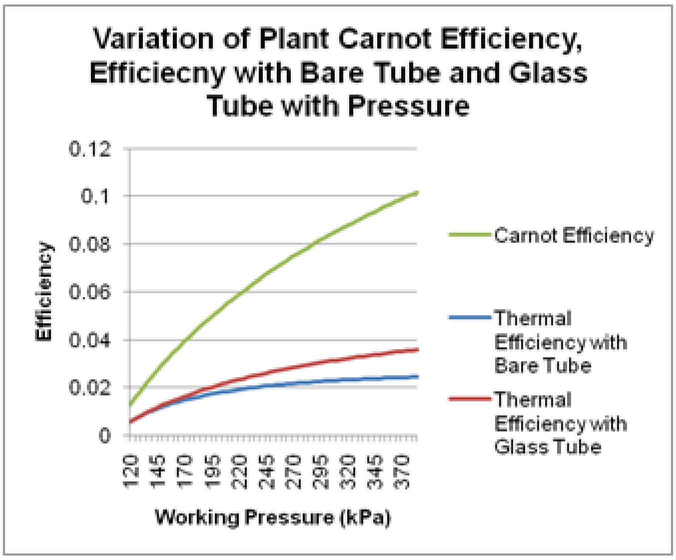

Figure 14: Plant Efficiency against Increasing Pressure

D. Mass flow rate

It is observed that by increasing pressure, the mass flow rate decreases for the same power output due to the increase in energy of the fluid producing the same power.

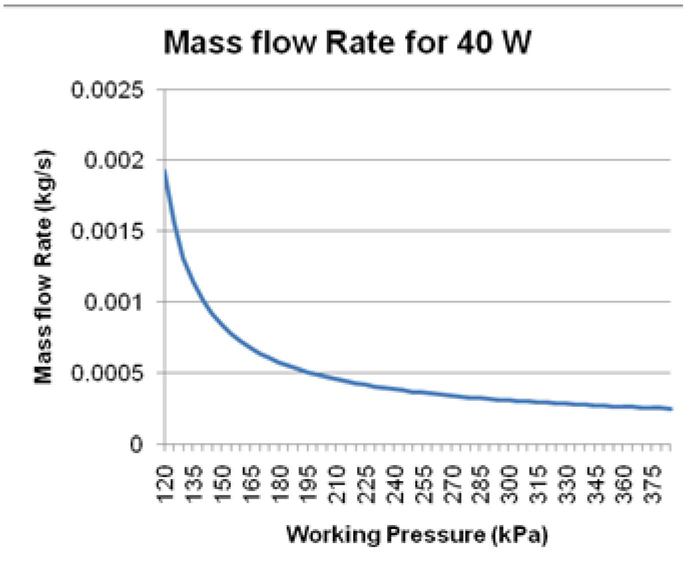

Figure 16: Mass flow Rate against Increasing Pressure

VII. Conclusion and future work

It is observed that plant performance improves at higher pressure. For 140kPa, 0.12m is optimized length of super heater, for uniform parabola width for entire boiler and super heater. Heat losses are reduced significantly withapplication of glass tube. Further losses can be reduced if this tube is evacuated. An

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experimental setup shall be established at Pakistan Navy Engineering College and these mathematical calculations will be validated in future work.

Acknowledgement

I am thankful to Cdr. Dr. Shafiq-ur-Rehman, Assistant Professor at Pakistan Navy Engineering College, NUST Karachi, for providing me guidance and helping me in design calculations for this Lab-Scale Solar Power Plant.

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